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Bioethanol E85 as a fuel for dual fuel diesel engine

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ABSTRACT

This study investigates the potential of E85 fuelling in a diesel engine. Researches were performed using a three-cylinder a direct injection diesel engine. A dual-fuelling technology is implemented such that E85 is introduced into the intake manifold using a port-fuel injector while diesel is injected directly into the cylinder. The primary aim of the study was to determine the operating parameters of the engine powered on E85 bioethanol fuel in dual fuel system. The parameters that were taken into account are: engine efficiency, indicated mean effective pressure, heat release rate, combustion duration and ignition delay, combustion phasing and exhaust toxicity. With E85 fuel participation, NO_x and soot emissions were reduced, whereas CO and HC emissions increased considerably. It was found that E85 participation in a combustible mixture reduced the excess air factor for the engine and this led to increased emissions of CO and HC, but decreased emissions of nitrogen oxides and soot.

1. Introduction

Apart from the economic issues, the extensive use of fossil fuels is responsible for a long-term environmental threat in the form of climatic changes and the slow (but continuous) increase in the average global temperature [1]. The most unfriendly to the environment is CO_2 which contributes to global warming. The European Parliament passed Directive 2009/28/EC on the promotion of the use of energy from renewable sources. This provision requires EU member states to use 10% of renewable fuels in transport by 2020. Biofuels made from agricultural products reduce the dependence of countries on oil imports, support local agricultural industries and enhance farming incomes [2].

Dual fuel technology. In practice, there are two ways of co-combustion of alcohol and diesel fuel. The first way is to produce a blend of diesel fuel with alcohol and then bring the mixture to the engine, using a typical supply system for a diesel engine (Fig. 1a). The greatest difficulties are that large percentages of alcohol do not mix with diesel fuel, hence use of diesel–alcohol blends is not feasible. Also, the blends are not stable and separate in the presence of trace amounts of water. In such a power system cannot change the ratio of diesel/alcohol.

The second way is fumigation system which injects an alcohol fuel into the intake port of an internal combustion engine (Fig. 1b). Into the engine cylinder is delivered air–fuel mixture, nearly homogeneous. The ignition process is controlled by the injected dose of diesel fuel. This requires the addition of a injector, along with a separate fuel tank, lines and controls.

Some advantages of fumigation:

- fumigation can substitute alcohol for diesel fuel; the some part of the fuel energy can be derived from alcohol by fumigation,
- the alcohol fuel system is separate from the diesel fuel system – the flexibility of power system; the engine can switch from dual fuel to diesel fuel operation. The diesel engine can operate with diesel fuel only [3];
- fumigated alcohol fuel tends to reduce smoke; this is particularly noticeable at high loads, where an engine is limited in power output due to smoke emissions [4–9];
- the alcohol injector is placed at the intake manifold – small modification of the engine; there is also a need, a simple of power and control system.

Ethanol is a biomass based renewable energy source, which can be produced with relatively low cost. Ethanol is characterized by high octane number. For this reason, ethanol is considered as a good fuel for spark ignition engines. Ethanol is considered as a fuel for compression ignition engine as well. In a diesel engine, the fuel alcohol is typically used in dual fuel system. Ethanol became established as an alternative fuel in1970s due to oil crisis [10,11].

The ethanol can be produced from crops, like corn, vegetables etc. Research continues on the development of high efficient, low cost processes for producing ethanol from other feedstocks such as waste from agricultural crops, food and beverage processing, wood and paper processing, and municipal refuse [12]. CO_2 is the

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Nomenclature

$(A/F)_{st}$	stoichiometric air fuel ratio	n	polytrophic index
BDC	bottom dead center	O	oxygen
C	carbon	p	pressure
CA	crank angle	TDC	top dead center
CO	carbon monoxide	THC	total hydrocarbons
CD	combustion duration	T	temperature
H	hydrogen	V	volume
HRR	heat release rate	φ	crank angle
N	nitrogen		
NO_x	nitrogen oxides		

main greenhouse gas and the CO_2 released by biofuel combustion can be fixed by growing plants and therefore makes no net contribution to global warming [13]. Ethanol is usually associated with spark-ignition engines because this fuel does not create difficulties for injection systems such engines. Due to higher octane rating of ethanol, engine knocking is not problematic and it can be burned in higher compression ratio. The problem could be water content in the ethanol which cause phase separation. Additional a higher heat of vaporization causes some ignition problem during cold start. These characteristics of ethanol does not matter when it is burned in the compression ignition engine. Hydrated ethanol can be directly used in diesel engines [14]. Diesel engine during warm-up period can operate on diesel fuel only to avoid disadvantage of cold start. In addition, ethanol is an oxygenated fuel and is likely to burn in a lean premixed mode, therefore helping to reduce harmful soot emissions that are problematic in conventional diesel engines [14]. Ethanol-only combustion in a diesel engine has great potential to achieve high efficiency low emissions combustion regimes such as homogenous charge compression ignition and stratified charge compression ignition [14–16]. The dual-fuel engines ware first developed to utilise natural gas. These studies report that dual-fuel combustion of premixed natural gas has advantages of increased efficiency and decreased smoke and nitrogen oxides (NO_x) emissions [14]. Dual fuel engines are also used for the combustion of generator gas [17,18] or other fuels of vegetable origin [19].

The studies deliver mixed conclusions on the maximum ethanol fraction for successful engine operation, some report the engine cannot run at over 15–20% ethanol fraction due to engine knocking [14] on the other hand others claim ethanol fractions of 60% or higher are acceptable. Chauhana et al. [20] presented fumigation system for introduction of ethanol in a small capacity Diesel engine and its effects on emission. Fumigation was achieved by using a constant volume carburetor. Different percentages of ethanol fumes with air were then introduced in the Diesel engine, under various load conditions. During the presented study, gaseous emission decreased with ethanol fumigation. Results showed that fumigated Diesel engine exhibit better engine performance with lower NO_x, CO, CO_2 and exhaust temperature. Ethanol fumigation has resulted in increase of unburned hydrocarbon (HC) [20]. Zhang et al. [21] presented the effects of fumigation methanol on the combustion and particulate emissions of a diesel engine under different engine loads and fumigation level. Fumigation methanol increases the peak heat release rate and ignition delay but does not significantly change the combustion duration. The fumigation method results in a significant decrease in particulate mass and number concentrations from medium to high engine loads, due to the increase of fuel burned in the premixed mode. Modeling researches are also performed on dual fuel engines. Gonca [22] at his work presented the results of model tests of the effects of steam injection on performance and NO emissions of a diesel engine running with ethanol–diesel blend. In his study, steam injection method was applied into a single cylinder, four-stroke, direct injection, naturally aspirated diesel engine fueled with ethanol–diesel blend in order improve the performance and NO_x emissions by using two-zone combustion model for 15% ethanol addition and 20% steam ratios at full load condition. He stated that the effective efficiency and effective power of the engine fueled mixture is increased up to 12.5% and 4.1%, respectively, NO emissions reduced up to 34% with steam injection.

Gasoline is also used to improve the operation parameters of diesel engine. There are works where fumigation gasoline is used to improve engine performance and reduce the toxicity of exhaust gases. Sahin and Durgun [23] recommended that gasoline fumigation can be effectively employed in existing diesel engine to improve engine performance and to reduce NO_x emission. Some studies of the dual-fuel combustion mode pointed out that using diesel to ignite gasoline mixtures in the cylinder can effectively suppress knocking and allow earlier ignition or a higher CR which results in a higher thermal efficiency [24]. During study of Labeckas and Slavinskas [25] was used a mixture of ethanol and gasoline to co-firing with vegetable oils. The result of these studies was that total NO_x emissions determined for the engine fuelled with various ethanol, gasoline and rapeseed oil blends depends on load, speed and the mass percentage of fuel bound oxygen [25]. Splitter and Reitz [26] was used E85 to power a diesel engine running in a dual fuel system. They noticed decreasing the gross thermal efficiency. They stated that losses can be minimized through proper balancing of the intake pressure and temperature, which are affected by fuel reactivity differences [26].

In the paper of Ma et al. [27] are presented results of the effects of diesel injection strategies on combustion, emissions, fuel economy and the operation range with high efficiency and low emissions fuelled with gasoline/diesel dual fuel on a modified single-cylinder diesel engine. This gasoline/diesel dual-fuel combustion mode proposes port fuel injection of gasoline and direct injection of diesel fuel with rapid in-cylinder fuel blending. The experimental results showed that this combustion mode had the capability of achieving high efficiency with low NO_x and soot emissions by using an early injection timing of single dose of diesel fuel. For high gasoline ratio the double injection strategy included the first and second injection timing was required [27].

In the present work as a fuel for dual fuel engine used E85 ethanol fuel. E85 is a high-level gasoline–ethanol blend containing 85% ethanol by volume. The addition of gasoline to ethanol (15%) is due to the need to improve the low temperature properties. Gasoline fulfills a role of improver fuel properties and facilitates cold starting SI engines. In the present study bioethanol fuel E85 was selected due to being market available ethanol fuel.

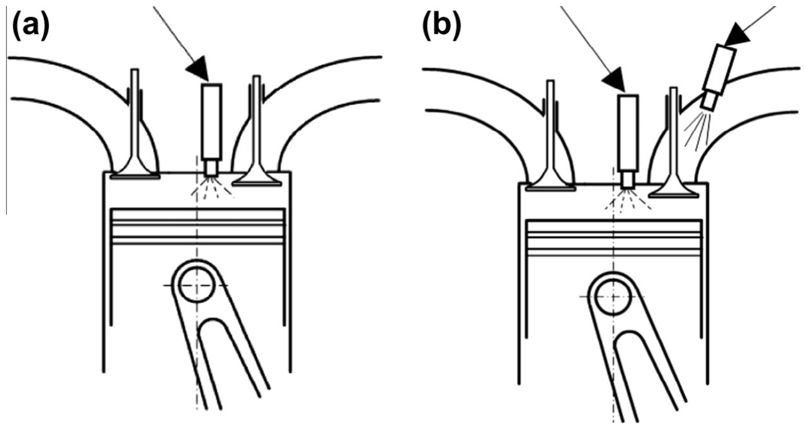

Fig. 1. Power systems of dual fuel engine, (a) supply by blend of fuels, supply by two separately systems (fumigation system).

Table 1
Properties of diesel fuel and E85, [28,29,30–32,25,33].

Properties	Diesel	E85
Molecular formula	$C_{14}H_{30}$	C_2H_5OH + gasoline (C_7H_{16}) or $C_nH_{2.88n}O$
Molecular weight	198.4	56.29
Cetane number	51	~11 (pure ethanol) 20–25 (gasoline)
Lower heating value, (MJ/kg)	41.66	29.6
Density at 20 °C (kg/m^3)	856	785
Viscosity at 20 °C (mPa s)	2,8	1.2
Heat of evaporation (kJ/kg)	260	780
Stoichiometric air fuel ratio $(A/F)_{st}$	14.7	9.92
Autoignition temperature (°C)	230	420 (pure ethanol) 300 (pure gasoline)
Flame speed (m/s)	0.86	~3
Flame temperature, $(A/F)_{st}$ (°C)	2054	2120 (ethanol)
Carbon content (wt%)	87	Ethanol 52.2, gasoline 85.5

Table 1 shows the parameters of the diesel and E85 fuel. E85 has a lower heating value compared to diesel. Another significant difference is 3 times larger heat of vaporization of E85 compared to diesel fuel. Ethanol as a main component E85 contains the fraction of oxygen, which results in lower stoichiometric air fuel ratio.

The energy percentage of E85 was calculated as follow:

Fig. 2a shows diesel and E85 masses for various E85 energy fractions used in the study. E85 is mainly bioethanol which contains a particle of oxygen. The molecule C_2H_5OH contains the oxygen. Supplying such fuel into the cylinder, oxygen is also provided indirectly [34]. The Fig. 2b shows the mass fraction of oxygen in the fuel (E85+diesel fuel) delivered to the cylinder. The first, purple bar represents the fuel supplied to the engine as a pure diesel fuel, without the E85 and without the presence of oxygen in the fuel. The fuel consumption rate with E85 was higher compared with diesel fuel because of lower calorific value of alcohol. Actual air–fuel ratio decreases with increasing E85 fraction. However, a stoichiometric air–fuel ratio of E85 is 9.92, significantly lower than diesel's 14.7. The decrease in both the stoichiometric air–fuel ratio and actual air–fuel ratio with increasing E85 energy fraction resulted in small variations in the overall excess air ratio. For example, when changing the E85 fraction from 0% to 90%, the excess air ratio changed from 1.69 to 1.82.

2. Test stand

Experiments were carried out on a generator set, which contains IVECO AIFO 8031 three-cylinder, naturally aspired direct injection diesel engine equipped with a MARELLI M8b 160 synchronous generator and the required control system. The synchronous speed of generator was 1500 rpm. The measured power and efficiency value were divided by the efficiency of the generator (which is 0.9 according to its manual) to achieve effective values. The engine specifications are listed in Table 2 and experimental setup is shown as a schematic in Fig. 3. The engine has 104 mm bore and 105 mm stroke with compression ratio 17:1. At each mode of operation, the engine was allowed to run for a few minutes until the cooling water and oil temperature have attained steady state values.

Data from the exhaust gas analyzer were recorded on a computer at intervals of one second. As a representative value,

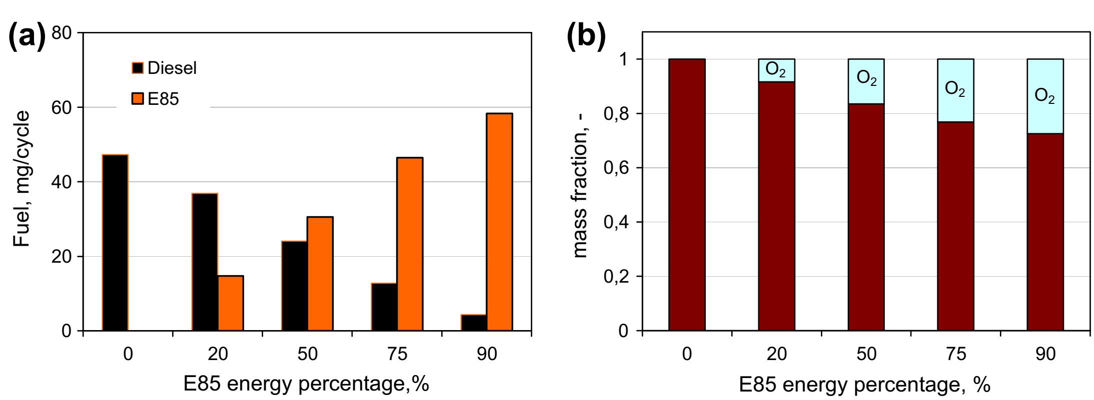

Fig. 2. The fuel doses by mass and oxygen content in ethanol fuel E85.

Table 2
Engine parameters.

Parameter	Value	Unit
Number of cylinders	3 in line	–
Displacement	2.9	dm^3
Bore	104	mm
Stroke	115	mm
Compression ratio	17:1	–
Rated power	24	kW
Rotational speed	1500	rpm
Angle of diesel fuel injection	7.5/5	deg BTDC

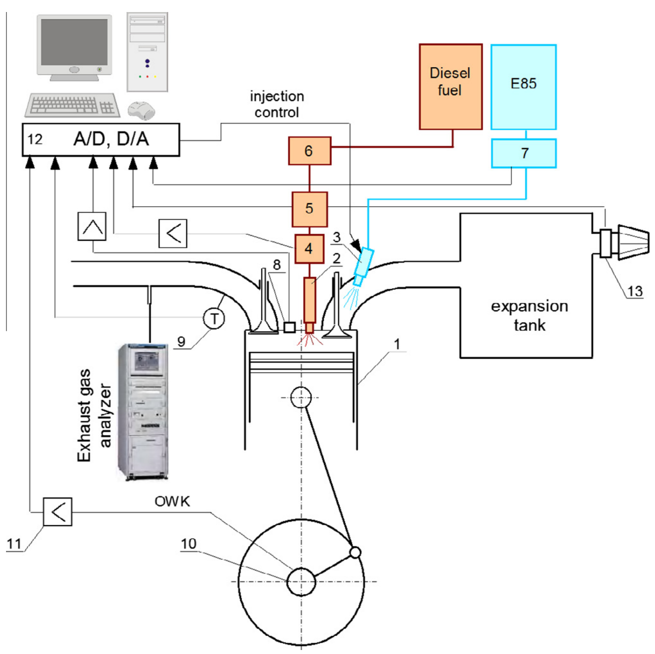

Fig. 3. Diagram of the experimental setup 1 – engine, 2 – injector, 3 – alcohol injector, 4 – fuel pressure sensor, 5 – fuel pump,6 – fuel flowmeter, 7 – alcohol consumption measurement system, 8 – pressure sensor, 9 – exhaust gas temperature sensor, 10 – crank angle sensor, 11 – amplifier, 12 – A/D, D/A module with PC computer, 13 – intake air flowmeter.

assumed the average value of data collected over three minutes of engine operation and the average results are presented.

The injection timing and duration were controlled using an electronic injection controller equipped with fuel pressure sensor. For E85 injection, the engine intake manifold was modified to install a port fuel injector. The injector was located close to the intake port so that the sprays are directed toward the valve top surface. The injection timing and duration were controlled using another electronic injection control unit. Power system ensures a constant pressure on the supply bus of at 5 bar using PID controller. The amount of fuel supplied was controlled by injector opening time. Fuel was injected into the manifold during the intake stroke. As the angle of start of diesel fuel injection for dual fuel engine, assumed the optimum angle for engine operating on pure diesel. During the study the pure diesel fuel without the addition of biocomponents was used. For the full load of engine angle of diesel fuel injection was 7.5 deg BTDC and at the partial loads 5 deg BTDC.

Measuring system:

- Diesel fuel consumption – AVL 7031 gravimetric meter, ±20 g/h
- E85 consumption – digital scale elapsed time, ±1 g
- Engine power – electrical power meter,
- Exhaust emissions (NO_x, CO, HC) – Horiba Motor Exhaust Gas Analyzer MEXA 8120F, Horiba Pre-Sampler, ±1% of full scale for each analyser module,
- Exhaust emissions (CO_2, O_2) – Sick Maihak S710 Gas Analyzer, $\leqslant$ 1% of selected output rang,
- Smoke meter AVL 415, ±3% of measured value,
- In-cylinder pressure sensor – Kistler 6001, sensitivity: ±0.5%,
- Charge amplifiers – Kistler 5001,
- Pilot injection timing – piezoelectric sensor attached to the high pressure fuel pipe – Kistler 6001 with adapter – Kistler 6501, sensitivity: ±0.5%,
- Resolution for the data acquisition system – 0.35 CA deg.

In cylinder pressure measurements were carried out using a piezoelectric pressure transducer (Kistler 6001). The HC and CO and NO_x emissions were measured using a non-dispersive infrared analyser (Horiba MEXA 812F) and CO_2 and O_2 was measured using Sick Maihak S710 Gas Analyzer. This instrument use heated sampling line. The smoke meter (AVL 415) was used to measure the exhaust smoke level. The intake air temperature was monitored during the engine tests and was around 24 °C. The total fuel energy per engine cycle was fixed and was about 1900 J. To increase E85 energy percentage, the diesel mass per engine cycle was decreased by reducing the injection duration while E85 mass was increased by extending the injection duration. These injection durations were determined by using measured injected mass per injection for diesel fuel.

3. Results and discussion

3.1. Calculations

In each engine operating point recorded 100 consecutive cycles. Each cycle was treated individually, for each cycle IMEP was determined. As a representative value of the IMEP, it was the average value from 100. Used own software, which calculates the engine cycle parameters based on the pressure course. Heat release rate (HRR) was calculated from the measured in-cylinder pressure data and crank angle readings. The basis of determining the heat release rate was the first law of thermodynamics and the equation of state. After rearranging and simplifications, the heat release rate vs. crank angle is obtained in well-known form as follows:

$$(\mathrm{HRR}) = \frac{dQ}{d\varphi} = \frac{1}{\gamma - 1}\left[\gamma p \frac{dV}{d\varphi} + V\frac{dp}{d\varphi}\right] \tag{1}$$

where γ – the ratio of specific heats, V – cylinder volume, p – in-cylinder pressure. Instantaneous cylinder volume V determined on the basis of the engine geometry.

Due to omitting as follows: heat transfer to walls, crevice volume, blow-by and the fuel injection effect, the resulted heat release rate is termed as the net heat release rate. The cumulative net heat released was obtained by integrating Eq. (1) over the crank angle φ.

The compression process in the internal combustion engines is considered as a polytrophic process where $pV^n = const$. The polytrophic index n can be computed from the slope of the curve on the $\log p - \log V$ coordinating system. The polytrophic index was determined with the equation:

$$n = \frac{\log p_2 - \log p_1}{\log V_1 - \log V_2} \tag{2}$$

where p_1, p_2 V_1, V_2 – pressures and volumes at the compression stroke, respectively.

Polytrophic index n was determined for each share of E85 fraction.

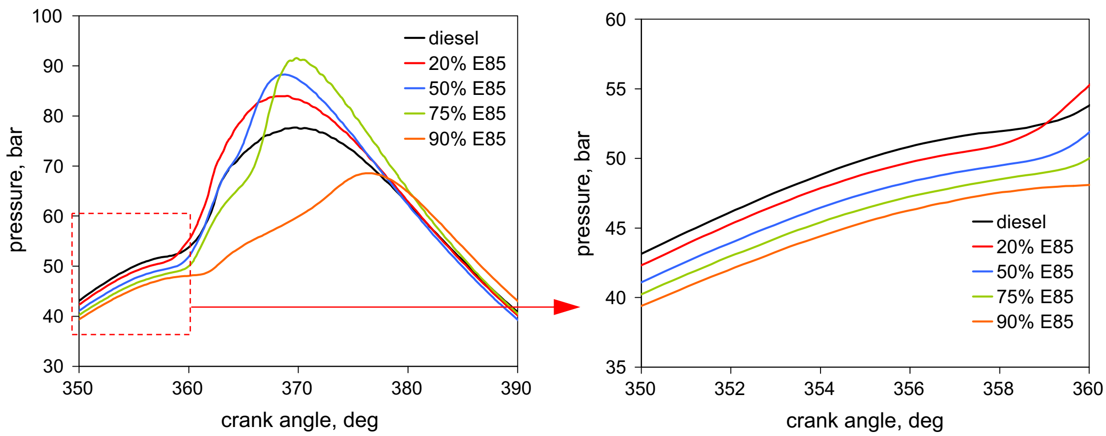

Fig. 4. Pressure traces for dual fuel engine at full load. Effect of E85 on the value of compression pressure.

3.2. Engine performance characteristics

Thermodynamic analysis of thermal cycle of dual fuel engine is based on the use of combustion pressure courses. For each operating condition, in cylinder pressure traces from 100 cycles were recorded. For each separate cycle were calculated thermodynamic parameters, and then were determined the value of the average parameters.

Fig. 4 shows the courses of pressure of dual fuel engine for various energetic shares of E85 and operating at full load equal 24 kW. It can be seen that an increase in the E85 fraction leads to a decrease in the in-cylinder pressures during compression stroke. This happens because of the vaporisation cooling of E85 fuel which is 3 times higher than diesel fuel (Table 1). It is causes decreased TDC pressure and temperature. On the basis of the analysis of the variations in pressure, it can be concluded that for the larger (50%) fraction in E85, the combustion process takes place in stages. A more detailed analysis of this phenomenon will be presented in the analysis of the heat release rate of the test engine.

Up to 75% of E85 energetic percentage, an increase in the value of maximum cylinder pressure was observed. For engine powered by diesel fuel, the value of pressure was equal to 77.5 bar. The value of maximum pressure was obtained for 75% share of the E85, and was equal to 91.5 bar and was higher by 14 bar relative to the only diesel powered engine. Cooling effect of E85 resulted in a reduction of in-cylinder pressure at the beginning of the injection of diesel fuel (7.5 deg BTDC) by 4.5 bar. With increase in the percentage of alcohol the polytrophic index decreases. It is associated with decreased of temperature (Fig. 5).

Fig. 5 shows the courses of temperature of dual fuel engine for various energetic shares of E85. The alcohol fuel E85 could down temperature of the fresh charge (air/fuel mixture) due to its much higher latent heat of evaporation than diesel fuel. The lower temperature of the charge, at the time of ignition, increases the ignition delay time. Evaporation of ethanol in the intake air (fumigation case) lowers the intake mixture temperature and increases it is density. Cooling effect of E85 resulted in a reduction of in-cylinder temperature at the beginning of the injection of diesel fuel (7.5 deg BTDC) by 60 K.

Fig. 6 shows pressure traces and heat release rates for various E85 energy fractions against crank angle degrees for full load of engine (24 kW). The diesel fuel start was set at 7.5°CA before top dead centre (BTDC). Using measured pressure traces, the apparent heat release rates (HRR) were calculated and shown in Fig. 6.

A trend is observed from the figures (Fig. 6) that with increasing E85 fraction, the start of combustion is delayed, the combustion phasing is retarded, and the peak HRR increases up to 75% of E85. At 90% share of E85 combustion process has slowed noticeably. This was because the lower ambient pressure and temperature due to vaporization cooling of E85 extended the diesel ignition delay period. In addition, the injected fuel does not meet with the air but with the E85–air mixture and oxygen concentration would be lower compared with the pure air [35]. A smaller concentration of oxygen in the surrounding of fuel drops increases

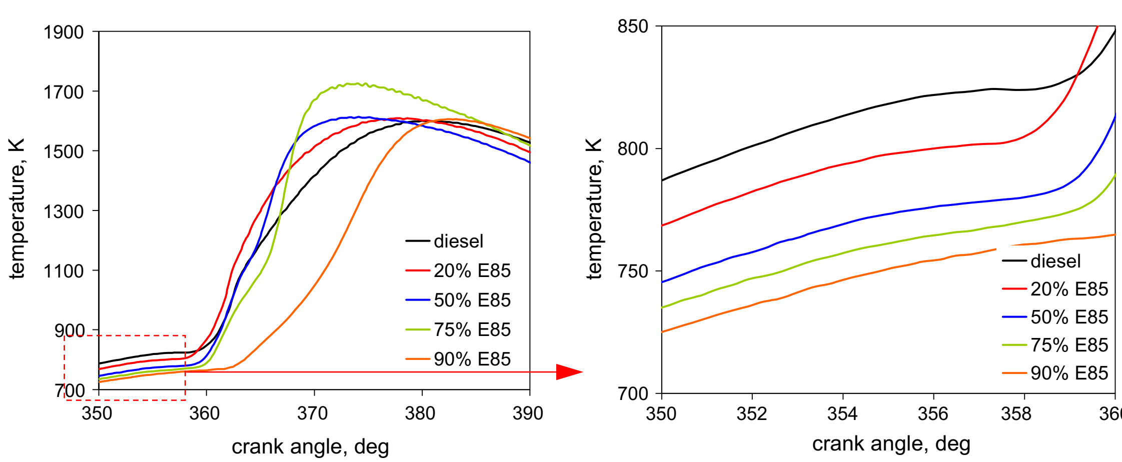

Fig. 5. Temperature traces for dual fuel engine.

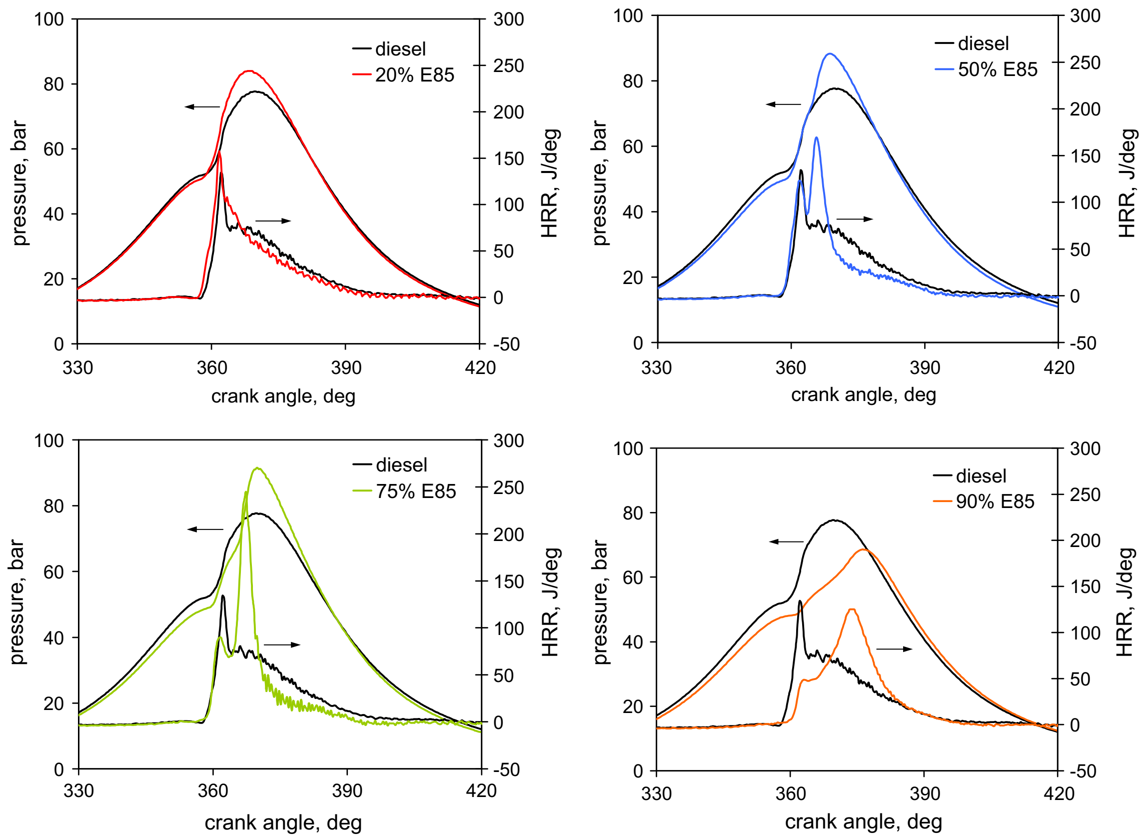

Fig. 6. In-cylinder pressure and apparent heat release rate for various E85 energy fractions. The diesel injection timing was fixed at 7.5 deg BTDC.

the ignition delay [36]. With the increase in the share of E85 is changing the nature of the course of HRR. Based on the data contained in Table 1, the auto-ignition temperature of diesel is nearly 200 K lower than the ignition temperature of ethanol and 70 K below the ignition temperature of gasoline. In the absence of diesel fuel ignition does not occur the ignition in the engine. There may be distinguished two phases of combustion. The first represents mainly the burning of diesel fuel and the second of E85. It is noticed that the combustion phasing is over-retarded for 90% of E85 leading to misfiring condition as the combustion occurs very late in the expansion stroke. This misfiring limits the maximum E85 fraction for this engine and operating condition to about 75% of E85. For 20% of the E85 participation, the nature of HRR was similar to that for engine powered by pure diesel. For larger proportions of E85 can distinguish two phases of combustion. The maximum value of HRR was obtained for 75% of the E85 and was equal to 245 J/deg.

The engine at full load noted the first signs of knock combustion, which limited the increase in the angle of injection diesel. At medium and low loads the knocking problem was not observed. The increased ignition delay extended time for pre-combustion mixing of the diesel fuel, which also results in an increased peak of HRR [36].

Fig. 7 shows the courses of changes in pressure and respectively HRR for partial loads. In both cases the angle of the start of diesel fuel injection is 5 deg BTDC. As with full load, reveals the influence of the cooling effect of E85 which creates a pressure drop at the end of the compression stroke. The shape of the courses of heat release rate was not different from obtained for the pure diesel fuel.

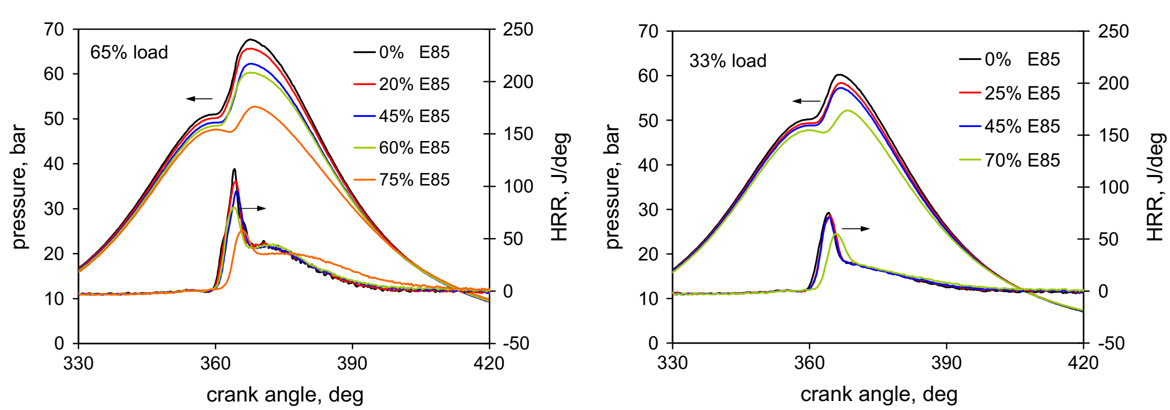

Fig. 7. Pressure and HRR courses at 65% and 33% load.

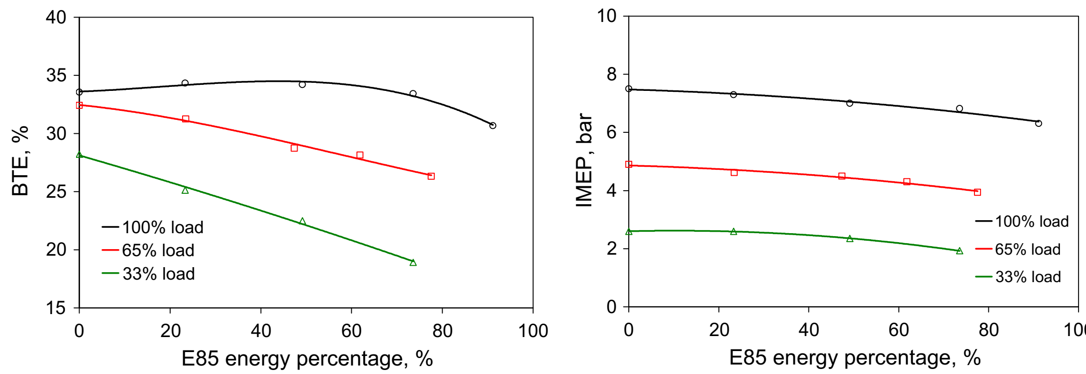

Fig. 8. Brake thermal efficiency and indicated mean effective pressure for various E85 energetic percentage.

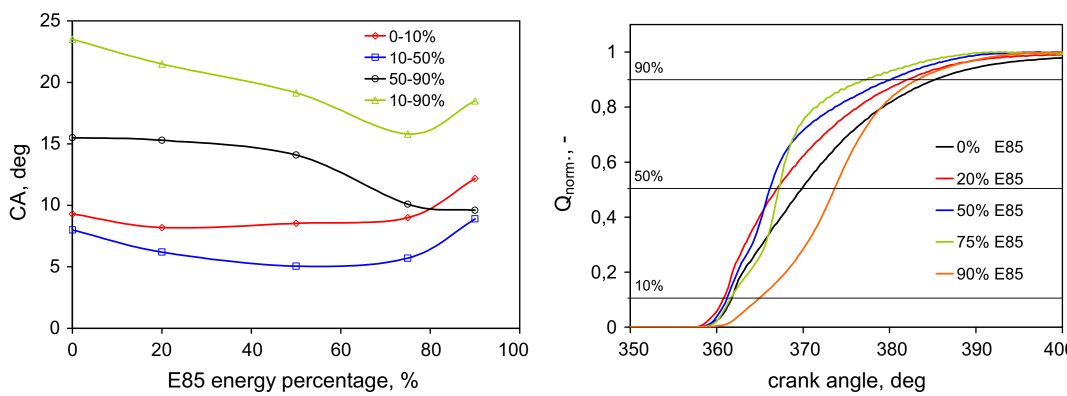

Fig. 9. Burn duration for various stages of combustion at different E85 energy fractions, full load (24 kW), injection time 7.5 deg BTDC and normalized cumulative heat release used to determine burn duration.

From the E85 energy variation, during tests at fixed diesel injection timing and full load of engine, it is found that increased E85 energy fraction increases the engine efficiency until the operation is limited by misfiring associated with over-retarded combustion phasing (Fig. 8). For partial loads with an increase in the share of E85 obtained a lower efficiency of the engine. By energy fraction, up to 75% of diesel was replaced by E85, efficiency gain compared with diesel-only operation was a little more than one percent higher. IMEP with increase in E85 fraction decreased slightly for all loads. In general, the slight gains in thermal efficiency with increased E85 (ethanol) substitution may be attributed to the increase in the ignition delay, so a rapid rate of energy is released which reduces the heat loss from the engine because there is not enough time for this heat to leave the cylinder through heat transfer to the coolant.

From the heat release traces, the ignition delay was calculated. The ignition delay is defined as the time between the start of diesel fuel injection and the crank angle of 10% heat release (CA 0–10%) [17]. Burn duration for the initial stage of the combustion is also calculated by reading the time between CA 0–10% and the crank angle of 50% heat release (CA 10–50%). Constantly, a difference between CA50 and CA90 is also calculated for the late-cycle burn duration. The ignition delay in diesel engine is defined as a time interval between the start of injection and start of combustion. This delay period consist of physical delay and chemical delay which occur simultaneously. In the physical delay takes place atomization, vaporization and mixing of air fuel, and in the chemical delay attributed pre-combustion reactions [37].

Flame propagation speeds and autoignition delay times are among the several fundamental combustion properties that depend on the fuel composition and structure. Autoignition delay time, in particular, is extremely sensitive to fuel type. Therefore, an understanding of the differences in the autoignition characteristics of various conventional and alternative jet fuels is necessary [37].

One of the significant factors in assessing the combustion process is ignition delay (ID) and combustion duration (CD). Shorter burn duration causes higher average rate of heat release. The ignition delay of dual fuel engine decreases with increasing E85 fuel fraction up to 75%. For the largest share of E85 the ID increased to a value of 9 deg and was only 1 deg higher than for pure diesel. The fastest initiate of the combustion process was noted for 50% of the E85 and was equal 5 deg of crank angle (CA).

To determine the combustion process next burn phase is interesting (CA 10–50%) where the heat release is driven mostly by the combustion of both diesel and E85 in premixed stage. Up to 75% E85 fraction this time was kept almost constant and was a bit smaller than for pure diesel fuel, increased for the largest share of E85. The late cycle combustion process (CA 50–90%) slowly decreased up to 50% of E85 fraction and decreased strongly with increasing E85 fraction. In this stage the diesel fuel combustion is limited by relatively slow mixing process. If one takes into account the total time of combustion (CA 10–90%), with an increase in the share of E85 fuel total burning time decreases up to 75% of E85 fraction. Shortening combustion process reduces heat loss of engine. Labeckas states that the fuel oxygen mass content reflects changes of the autoignition delay time caused by the use of ethanol–diesel–biodiesel blends more predictably than the cetane number does [6].

The start of combustion is difficult to define. During compression stroke, diesel fuel of small droplets form is injected into the cylinder having air–fuel charge at high temperature [38].

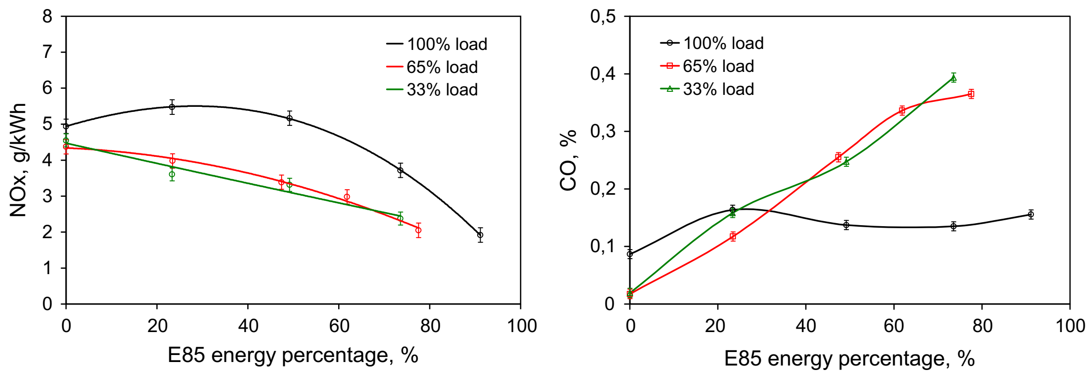

Fig. 10. Emissions of nitrogen oxides and carbon oxide for various E85 fraction and loads.

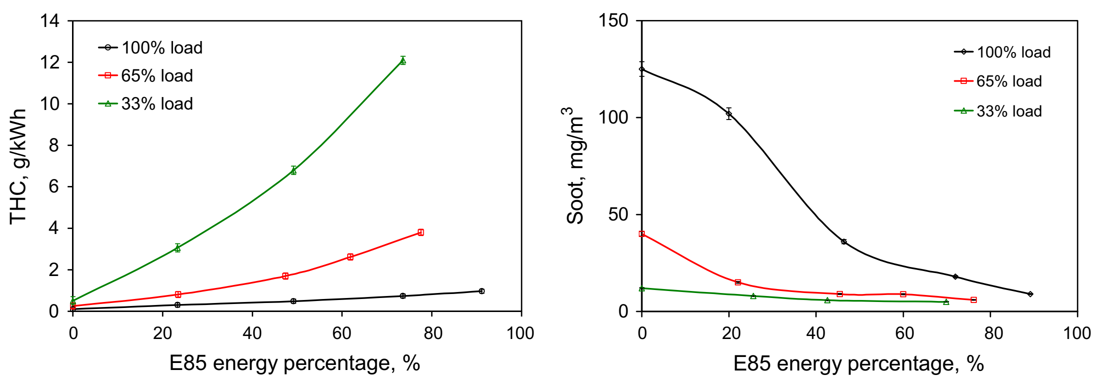

Fig. 11. Emissions of total hydrocarbons and soot for various E85 fraction and loads.

Based on the literatures review [39], it can be stated that alcohol fumigation in a diesel engine affects the brake thermal efficiency in two ways. Alcohol fumigation decreases the BTE at lower engine load condition and increases the BTE at higher engine load condition. At low engine loads, the excess air ratio is very high hence the intake air and the fumigation alcohol form a mixture which might be too lean to support combustion, resulting in deterioration of combustion efficiency and thus reduced the BTE. Alcohol has much higher heat of vaporization (780 kJ/kg) compared with that of diesel (260 kJ/kg). Due to this characteristic alcohol might cool down the combustible mixture hence there will be a drop in BTE. The increase of BTE at higher engine loads can be explained by the fact that homogeneous air/alcohol mixture burns faster hence provides more premixed combustion which tends to increase the BTE. Alcohol has lower cetane number which increases the ignition delay hence energy is released within a very short time, resulting reduction in the heat loss from the engine as there is no sufficient time for transferring heat through the cylinder wall to the coolant [39] (Fig. 9).

4. Emission

The HC, CO, NO_x and soot emissions were measured and CO_2 and O_2 ware measured as well. Fig. 10 shows the NO_x and CO emission of dual fuel engine for various loads. The NO_x formation in a diesel engine is depended on the combustion temperature and oxygen concentration in the combustion process.

In dual fuel engine different mechanisms are involved. Due to the higher latent heat of evaporation of E85 fuel could reduce the combustion temperature and consequently lead to reduction of NO_x. This mechanism dominates at partial loads, where lower combustion temperature and higher air/fuel ratio results in significant reduction of NO_x emission. For a full engine load and for 20% and 50% share of E85 an increase in the emission of NO_x was noticed. A further increase of E85 energetic share resulted in the reduction of NO_x. The diesel fuel is combusted in mixture of air and E85 fuel. The formation of NO_x is affected by the peak flame temperature and the content of nitrogen and oxygen available in the reactive combustible mixture at combustion zone. The oxygenated fuel E85 may provide additional oxygen for the formation of NO_x. On the other hand, the cooling effects of E85 can combustion temperature and NO_x emission.

The increase of CO affects, among others incomplete combustion (lower combustion temperature), poor mixing (local air/fuel equivalence ratio). In dual fuel engine the E85 fuel is burned as a homogenous charge. The flame has to propagate through the charge inside combustion chamber of engine. At partial engine loads, the cooling effect can lead to incomplete oxidation of the CO to CO_2 during the expansion stroke [40].

Fig. 11 shows emission of THC and soot. The THC and CO emissions in general decrease with increase in engine load, due to higher in-cylinder gas temperature.

Soot emission reduces with increase E85 share. There are several reasons to the lower soot emission from dual fuel engine. Less diesel fuel consumption because some part of the fuel is replaced by E85 fuel. This causes less diesel fuel takes part in the diffusion phase of combustion process. Due to longer ignition delay, more diesel fuel is burned in the premixed phase [41]. The diesel fuel consists of various hydrocarbons and the ratio of carbon to hydrogen (C/H) is high, and there is a tendency to form soot under fuel reach combustion conditions. E84 is characterized by lower C/H ratio and contribute to lower soot emission despite the fact that more mass of fuel is burned [42]. During the final stage of

compression stroke and combustion process, the air/E85 mixture trapped into crevices or quenched by the cylinder wall. Some part of this mixture can be adsorbed by oil film and can form unburned HC. Almost all studies report benefits of ethanol substituting diesel in reducing smoke emissions because ethanol is oxygenated fuel and the ethanol–air charge is well mixed [43].

Factors causing combustion deterioration (such as high latent heats of vaporization) could be responsible for the increased CO production. Combustion temperatures may have had a significant effect. A thickened quench layer created by the cooling effect of vaporizing alcohol could have played a major role in the increased CO production [44]. In case of CI engine powered by ethanol–diesel blend the increase of HC emission is noticed as well [4,45].

5. Conclusions

Dual-fuel combustion using E85 fuel port injection and diesel in-cylinder direct injection was tested in a three-cylinder compression ignition engine at various loads. The study of dual fuel engine was conducted for a constant angle of diesel fuel injection that was optimal for engine powered by diesel. The effect of E85 energetic percentage on the engine mean effective pressure and efficiency was analyzed. The impact of E85 energy fraction on emissions was presented and discussed. By using E85 with diesel fuel it can supply additional oxygen for the combustion which can cause improvements in overall combustion process. The combustion process in dual fuel engine takes place in a shorter time than in the diesel engine. Based on the tests it is difficult to draw conclusions about the impact of the gasoline contained in E85 on the engine operating parameters. There are difficult to separate the effect of ethanol on the impact of gasoline on dual fuel engine parameters. The major findings from this study are summarized as follows:

– with the application of E85, there is an increase in the total mass of fuel consumed while there is a reduction of diesel fuel consumed,
– with the application of E85, the maximum in-cylinder pressure decreases from low to medium engine load but increases at high engine load. For full load, with the participation of E85, the combustion process occurs faster than for the pure diesel. For example, for 50% of the E85 share, the burning time of 90% of fuel take 19 deg CA, but for 65% load and this same E85 share it was equal 31 deg CA, for 33% of load it was 32 deg. At partial loads combustion takes place with a higher excess air ratio. So with decreasing engine load, fuel burn time was lengthened,
– emissions of the unburned hydrocarbon (HC) and carbon monoxide (CO) increase and the BTE decreases with increasing E85 fraction. It can be explained that some ethanol is trapped inside the crevice resulting in increased HC emissions,
– share of E85 can effectively reduce soot emission. The reductions are due to the reduction of diesel fuel consumed as well as the increase in fraction of heat release in the premixed phase of combustion. Consistent with other studies, the soot emissions were largely dependent on oxygen content in the fuel. E85 showed excellent ability to eliminate smoke emissions.

Tests were conducted at a constant angle of diesel fuel injection. Optimization of the angle of injection would result in a smaller decrease in efficiency. For each E85/diesel percentage, engine control system should provide an optimum angle of diesel fuel injection. The elevated HC emission can be easily aftertreated by means of conventional oxidation catalytic converter. In summary, available for purchase E85 fuel can be used to power a diesel engine in dual fuel operation.

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